

## POSITIVE LUBRICATION SYSTEM

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by

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## ABSTRACT

As part of the development of an autonomous lubrication system for spin bearings, a system was developed to deliver oil to grease-lubricated bearings upon demand. This positive oil delivery system (PLUS) consists of a pressurized reservoir with a built-in solenoid valve that delivers a predictable quantity of oil to the spin bearing through a system of stainless steel tubes. Considerable testing has been performed on the PLUS to characterize its performance and verify its effectiveness, along with qualifying it for flight. Additional development is underway that will lead to the fully autonomous active lubrication system.

## INTRODUCTION

The useful life of a control moment gyroscope (CMG), reaction wheels, and momentum wheels is strongly dependent upon the spin bearing lubrication system. Current lubrication systems are passive in nature and consist of either a grease system or a controlled-leak-type oil system. The grease system has proven effective for lifetimes of up to three years for CMGs and up to 11 years in momentum wheels; systems have been designed for up to eight years in small CMGs. The controlled-leak-type, oil-only system has demonstrated encouraging performance in life tests but has had little experience in space and none on very active CMGs.

Future requirements for CMGs, reaction wheels, and momentum wheels include longer life and more rigorous service. Current data indicate that these increased demands are likely to stress the grease system beyond its endurance, and the passive controlled-leak oil system has questionable performance even for current demands. An alternative is an active oil system. Such a system would supply oil based on bearing requirements. The ultimate system would sense the oil requirement of the bearing and supply just enough oil to meet this need. This system would use a film of grease as a buffer, effectively storing any excess oil added to the bearing and metering the oil out to the running track when conditions demand it. Such a system could also reduce drag torque and unusual low-temperature runup behavior associated with excess lubricant within the bearing, while providing adequate lubrication for good reliability.

Progress toward an active autonomous oil lubrication system was initiated in 1983 when design was completed and testing started on a sensor that detects the ratio of cage rotational speed to shaft speed, which is a function of oil film thickness. In 1986, work was initiated on the oil pump and insertion system. This activity culminated in a positive lubrication system (PLUS) that will be used to supplement the life of the grease system by manually injecting oil at prescribed intervals. The PLUS will be available if needed because of unexpected circumstances.

## OBJECTIVE

The objective of the PLUS is to, on demand, inject Coray 100® lubricating oil into the spin bearings of a CMG, reaction wheel, or momentum wheel. This will supplement the present passive Andok C® grease lubrication system, increasing the life of the system.

## SUMMARY

The lubrication system currently being used consists of Andok C grease, which is initially packed between the balls but forms a channel as the bearing is run. The significant lubrication of the bearing is accomplished by the oil, which is supplied by the grease to the ball-race interface. In addition to grease along the ball track, some grease attaches itself to the cage and supplies oil to the ball-cage interface.

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There are two major mechanisms for loss of lubrication from the ball tracks. The first is through migration of the oil away from the bearing and the other loss is evaporation. Eventually in the life of any bearing system, the supply of oil available to the ball-race interface becomes inadequate. When this occurs, the drag torque and subsequently bearing temperature increase, which accelerates the lubricant loss and degrades the lubricant. This can cause cage instability, which accelerates the degradation process. Such behavior is an indication of the end of life (defined as excessive drag torque). A passive lubrication system (greased-packed, controlled-leak, etc) cannot correct these conditions when they occur. The PLUS allows periodic bearing relubrication to extend the life of the system or intervention to correct premature (prior to normal end of life) lubricant-related problems as they occur. The PLUS has the capability to replenish the oil in the bearing (on command) by directly inserting the oil onto the balls for immediate restoration of EHD film thickness and return to normal operation.

## **PLUS DESIGN CONFIGURATION**

The PLUS design consists of a pressurized reservoir with a solenoid valve, supplying oil through feed tubes to the bearings upon manual command.

### **PRESSURIZED RESERVOIR**

The main component of the PLUS is a reservoir/solenoid valve, shown in Figures 1 and 2. The reservoir is mechanically pressurized by a spring compressed between the bellows and the outer housing. A number of pump designs were initially considered for the PLUS during the development period. The most significant of these were peristaltic, piezoelectric, and pressurized reservoir designs. The peristaltic design was abandoned because of material concerns, and the piezoelectric design proved too sensitive to lubricant properties and environmental conditions. The following is a description of the pressurized reservoir valve design that was eventually adopted.

The amount of spring compression (i.e., reservoir pressure) is adjusted by the initial position of the outer housing, which is threaded onto the valve housing. The bellows is extended during the oil-fill process and provides additional mechanical pressure. The installed dimensions of the spring and bellows provide an initial reservoir pressure of 80 psia with a 25-to-35-psia pressure drop from full to empty. In the event of spring failure, the bellows alone can provide enough pressure to deliver oil. The reservoir is designed for a deliverable volume of three milliliters but more or less is easily accommodated with simple design changes. Oil is discharged by opening the valve for 125 milliseconds at 10-second intervals. The amount of oil discharged per pulse can vary from 0.2 to 5.0 milligrams per pulse depending on reservoir pressure, operating temperature, and plumbing flow resistance.

The 28-V dc solenoid valve (Kaiser Eckel), Figure 3, has an operating pressure of 100 psig and a maximum leakage rate of  $5.0 \times 10^{-6}$  standard cubic centimeters per second (SCCS) GHe. The valve is constructed of 316 CRES and has a Viton seat. The valve and the bellows are electron-beam welded into position and then helium leak tested.

The oil volume requirement for the pressurized reservoir was based on adding 20 to 30 milligrams/month for three years plus approximately 200 milligrams for initial system fill. The volume of three milliliters (2.8 grams) is more than a factor of two greater than the estimated oil required.

### **FEED TUBE/BEARINGS**

The testing and selection of the oil injection method is discussed in detail in the Oil Insertion/Transfer Design Section. A typical flight installation is shown in Figure 4. Oil is transferred from the valve to the delivery port in the bearing cartridge through a 0.063-inch outer diameter (OD) seamless CRES tube with a 0.016-inch inner diameter (ID). The actual design of the feed tube will vary considerably depending on the application and the oil insertion options. However, this size was selected because it was small enough to minimize the tube volume but large enough that small amounts of contaminants in the oil would not obstruct flow. Also, testing showed that this size feed tube functioned well over a wide range of operating conditions.

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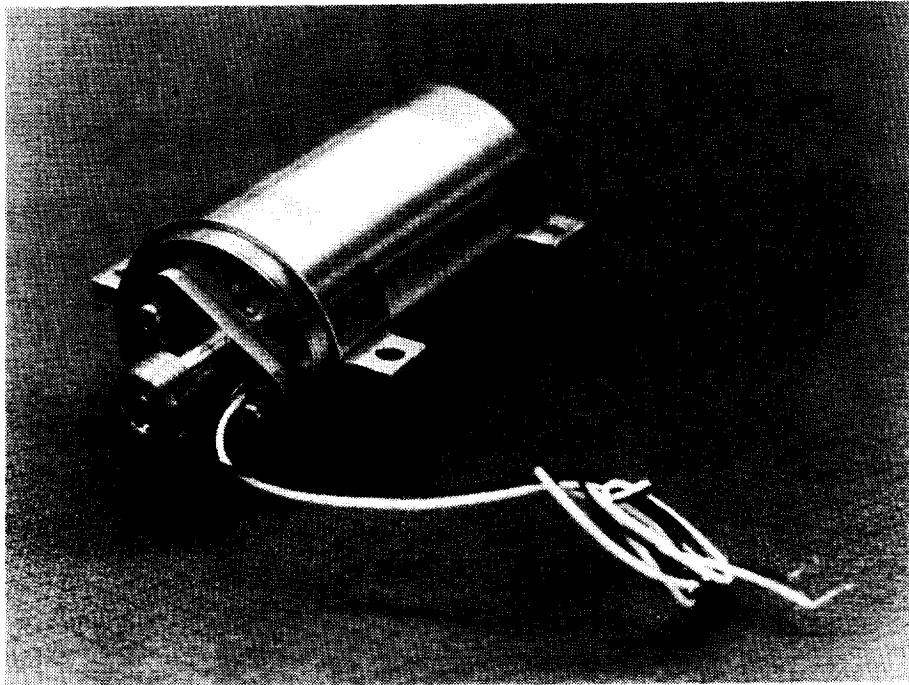


Figure 1. Reservoir Valve Assembly

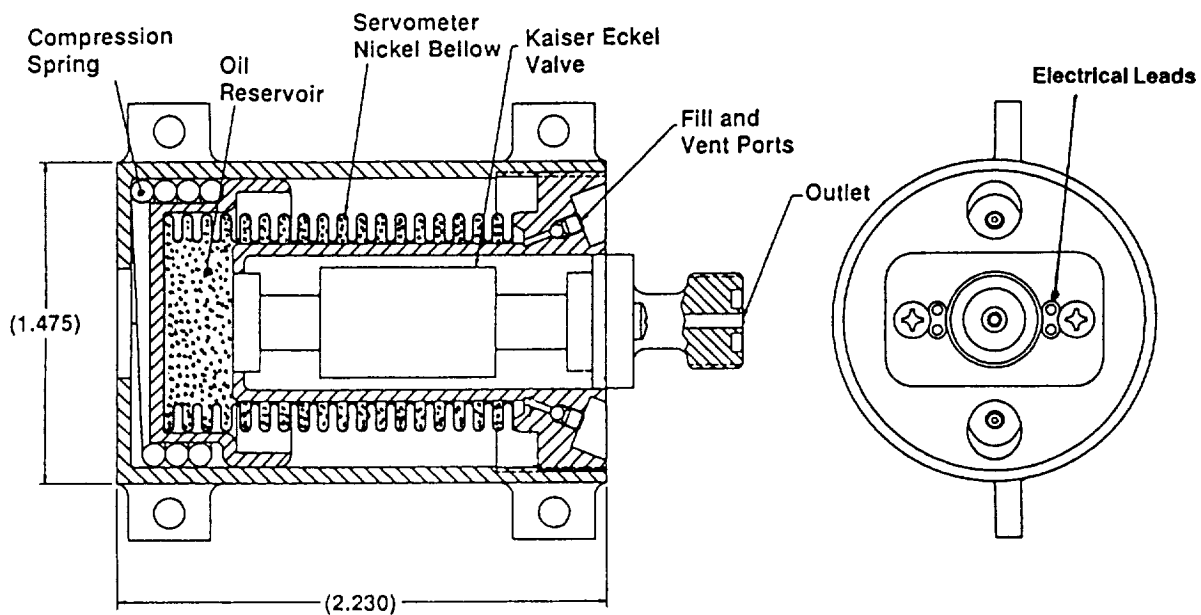


Figure 2. Pressurized Reservoir/Valve

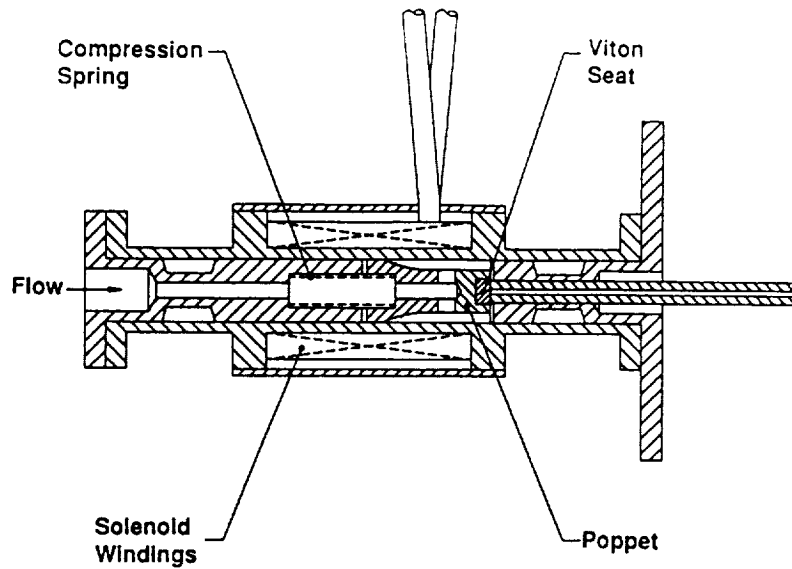


Figure 3. Kaiser-Eckel Solenoid Valve

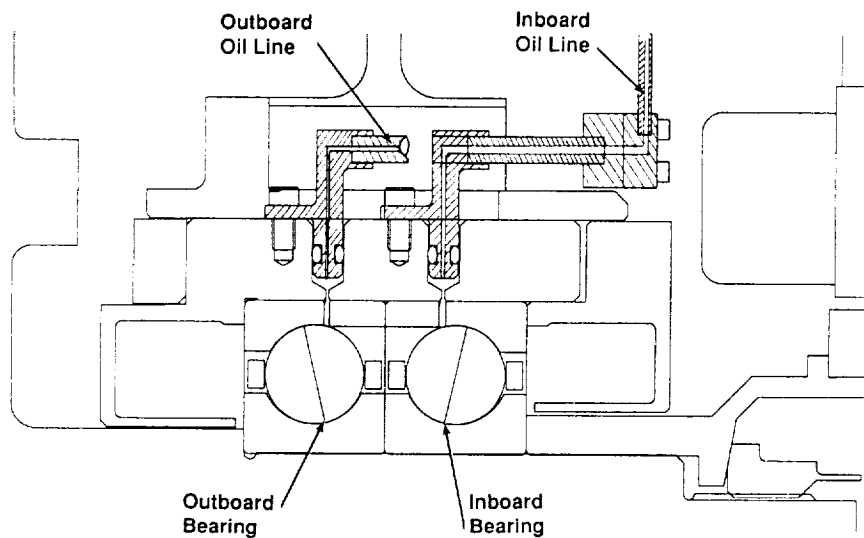


Figure 4. Typical Flight Installation

The bearing cartridge has two O-ring connector ports machined on the outer diameter to connect with the feed tubes. A 0.016-inch diameter hole connects these ports to the interior of the cartridge at a point that aligns with the center of a machined groove on the ID of the cartridge. This groove aligns with the four radial holes in the spin bearing outer races.

The spin bearings have four 0.016-inch OD radial holes in the each bearing 90 degrees apart. The holes go from the OD of the outer race to the interior of the bearing at a point within 0.004 inch of the balls.

## CONTROL/TELEMETRY

Although normal operation of the PLUS is injecting a fixed amount of oil at predetermined intervals, it will also be used in the event of anomalous behavior. Current telemetry (spin motor current command and bearing cartridge thermistor) is monitored to detect abnormal bearing behavior. Abnormal bearing behavior is generally characterized by an increase in drag torque (proportional to the spin motor current command) and a corresponding increase in bearing temperature. At the onset of these abnormal conditions, a signal would be sent to the PLUS to inject 20 to 40 milligrams of oil per bearing. Previous testing has shown that this quantity of oil will reduce the drag torque to normal levels.

In the present system, there is no direct measurement to determine the exact amount discharged from the PLUS after the system has been activated (pulsed). The amount discharged is calculated based on component test data, in-flight usage history, and operating temperature. An indicator is presently being developed (i.e., linear potentiometer) that will monitor bellows position and, with telemetry, will give a direct measurement of the amount of oil discharged as well as the amount of oil remaining in the reservoir.

## SYSTEM DESIGN DESCRIPTION

The present PLUS has two pressurized reservoir/solenoid valve assemblies per bearing for redundancy. When the command to inject oil into either the fixed or floating end is received, one solenoid valve per bearing opens and oil flows through the feed tubes into the small annulus between the outer races of the bearings and the cartridge (Figure 4). The oil then flows through the four small holes in the outer ring of each bearing from the annulus to the interior of the bearing near the balls. The PLUS design provides redundancy for both the oil supply and the oil paths.

## SUMMARY OF TESTS

### CHARACTERIZATION BENCH TESTING

Numerous tests were performed on the pressurized reservoirs to determine their operational characteristics. These tests determine the unit's output as a function of valve open/close time, temperature, and amount of oil previously discharged (i.e., reservoir pressure). This information is used during actual operation to determine the number of pulses required for a desired oil output.

**TEST SETUP** – The pressurized reservoirs used for the initial bench testing were flight-sized prototype units almost identical to the flight units shown in Figures 1 and 2, with pressure transducers installed to monitor oil pressure. These units were connected to various length/diameter feed tubes. The method of testing was to cycle the solenoid valve at various on/off times for a number of pulses while collecting and weighing the oil discharged and monitoring reservoir pressure. Testing was performed in both vacuum (bell jar) and atmospheric environments at various temperatures. The reservoir would be set at some initial pressure, which would slowly decrease as oil was removed from the reservoir.

**OBJECTIVES** – The objectives of this initial feed-tube/reservoir characterization testing using the prototype flight units were to determine the effects of various lengths and diameter of feed tubes/orifices, reservoir pressure, and temperature on oil delivery performance. Additional objectives were to define the performance and interface requirements of a flight PLUS.

**FEED-TUBE CHARACTERIZATION** – Numerous tests were performed on the pressurized reservoir with a variety of feed-tube configurations. The basic approach was to test

various length (3 to 12 inches) and ID (0.007 to 0.053 inch) feed tubes. Testing was conducted at temperatures between 20° to 120°F. Solenoid activation (pulse) time was varied between 25 and 5000 milliseconds; intervals between pulses were varied to determine the effects, if any, on output and to collect data required to optimize actual operation.

An example of the results of the testing is shown in Figure 5. The effects of reservoir pressure and temperature on oil delivery is shown on the upper plot. There is an increasing relationship between oil output and temperature that is due to an increase in pressure and a decrease in oil viscosity. The bottom plot shows the increased output as a function of pulse (valve open) time and temperature. The difference in oil output, shown on the top and bottom plots, is due to the difference in ID and length of the output feed tubes.

After the first-application flight design was selected, additional testing was performed with the prototype units to determine the performance with an even longer (>20 inches) feed tube. Although the calculated pressure drop showed adequate performance, a test was performed using the baseline-sized feed tube formed into a 2-inch diameter coil using about 20 inches of tubing. The testing was performed using a 2-milliliter capacity reservoir pressurized to 80 psia. The unit was pulsed 1700 times using 125-msec pulses at 10-second intervals. This test was conducted at temperatures from 20 to 75°F. The results of these tests are shown on Table 1. Although the output dropped considerably at 20°F (as expected), the pump/coil combination successfully delivered oil over a large range of temperature and reservoir pressure.

**SUMMARY OF RESULTS** – The performance testing demonstrated the ability of the units to perform under a variety of conditions. The performance of all units tested was uniform with consistent output. The testing also determined the performance characteristics of the system that are needed for actual operation during flight or future life simulation test.

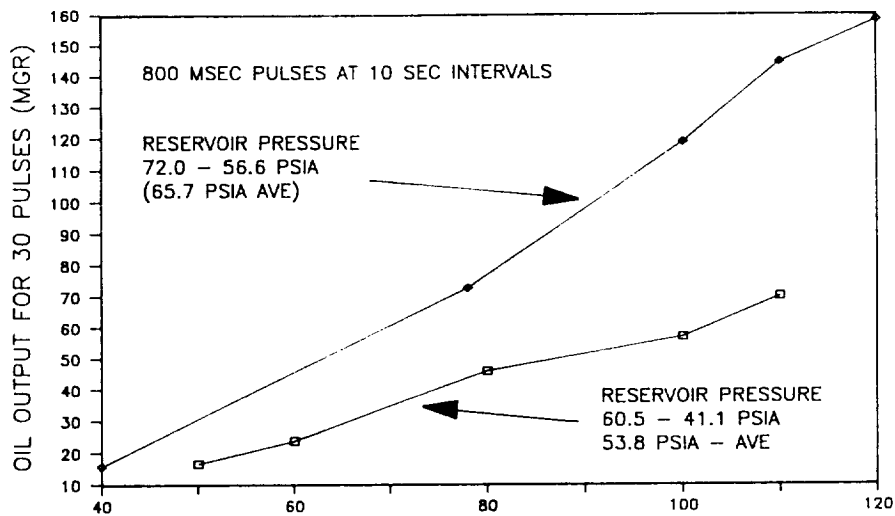
#### **BEARING TEST FIXTURE TESTING**

**OBJECTIVES** – There were two primary goals of the testing: to develop a method of transferring oil from the pressurized reservoir to the bearings and to determine if oil injected into the outboard bearing would transfer to the inboard bearing.

**TEST SETUP** – The test fixture used is shown in Figure 6. The drive system consists of a 65-in.-oz Kollemorgan Brush Motor/Tachometer with feedback control to maintain continuous 6000 rpm. A bellows coupling connected the Kollemorgan drive motor to a ferrofluidic feedthrough (made by Ferrofluidics Corp), which maintains the vacuum chamber's integrity. These components are external to the vacuum chamber.

The bearing cartridges and clamp rings for the tests are duplicates of flight design cartridges. The bearing cartridge was attached to a 50-in.-oz Lebow torque transducer to measure the drag torque. The cartridge is also equipped with a flight thermistor. Both the temperature and the drag torque are continuously monitored on a strip-chart recorder.

**OIL INSERTION/TRANSFER DESIGN** – Two areas of concern were how to insert oil into the bearings and whether a PLUS was needed for each bearing or just for the outboard bearing. For PLUS installation, the effort would be minimized if oil inserted into the outboard bearing would transfer to the inboard bearing. There were several methods tested for injecting oil into the bearings. The design of each method is described in the following sections.



OIL OUTPUT VS TEMPERATURE FOR VARYING PULSE TIMES  
ALL PULSES AT 10 SEC INTERVALS

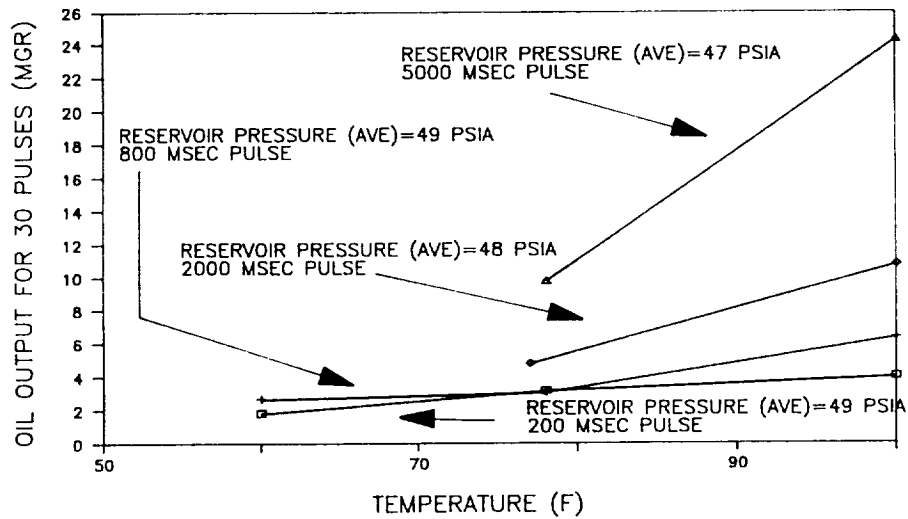


Figure 5. Oil Output versus Temperature

Table 1. Pump/Coil Performance Testing

Temp (°F)	No. of Pulses to Fill Coil (95 Mgrams)	Starting Pressure (psia)	Pressure Drop During Fill (psia)	Total Performance (Mgram/Stroke)	Total Pressure Drop (psia)	Average Pressure (psia)
75	35	80	7	1619/1700 = 0.95	35	62
60	40-45	76	3	601/443 = 1.36	10	71
40	60-70	64	4	408/465 = 0.88	10	59
20	200	62	1	102/600 = 0.17	9	57

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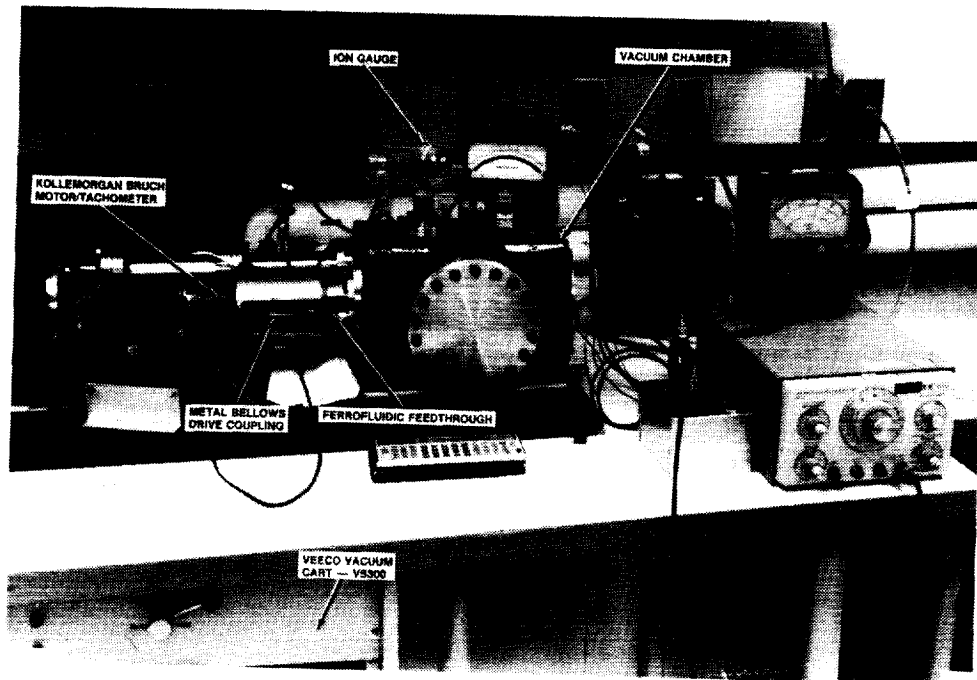


Figure 6. Oil Insertion Test Fixture

**INNER-RACE-TO-CAGE TRANSFER (OPTION 1)** – Option 1 inserted oil from a feed tube directly onto the spinning bearing inner race where the centrifugal force moved the oil to the underside of a lip on the bearing cage that protruded axially from the bearing (Figure 7). This cage with the designed lip was tested extensively with the PLUS in bench spindle testings but was never flight qualified. Oil insertion using this cage was demonstrated successfully, but the performance characteristics of this new cage were unproven. Due to the qualification time and departure from flight heritage, additional design and testing efforts were undertaken to develop a new method for inserting oil using existing flight-qualified cages.

**INNER-RACE/OUTER-RACE DEFLECTION RING (OPTION 2)** – Option 2 (Figure 8) used a feed tube to insert oil onto the spinning inner race. The oil would be centrifugally transferred to an oil deflection ring that would direct the oil into the outer race and balls. During the testing of Option 2, the cage (retainer) occasionally contacted the oil deflection ring. Based on this data and the unknown long-term effects that the ring could have on grease-cage coupling, the decision was made to eliminate this option.

**DIRECT-TO-BALLS/CAGE-OUTER-RACE INSERTION (OPTION 3)** – Option 3 (Figure 9) used a feed tube inserted between the cage and the bearing outer race to deliver oil directly onto the balls. The feed tube extending into the bearing was 0.025 inch thick. The cage-to-outer-land radial clearance was calculated to be 0.020 to 0.025 inch. With this tight clearance, there was concern that the cage might contact the lip if there was wear on the cage ID. It was decided that the flight design feed-tube lip would be made into a complete ring to provide a smooth surface identical to the outer land. A detailed analysis was performed to determine the actual cage-to-outer-race clearance. If this option was to be used, the bearings would have to have 0.025 inch of material removed from the ID of the outer land in order to maintain the existing cage radial clearance. An analysis showed that this material removal would not prevent the outer land from supporting design loads.



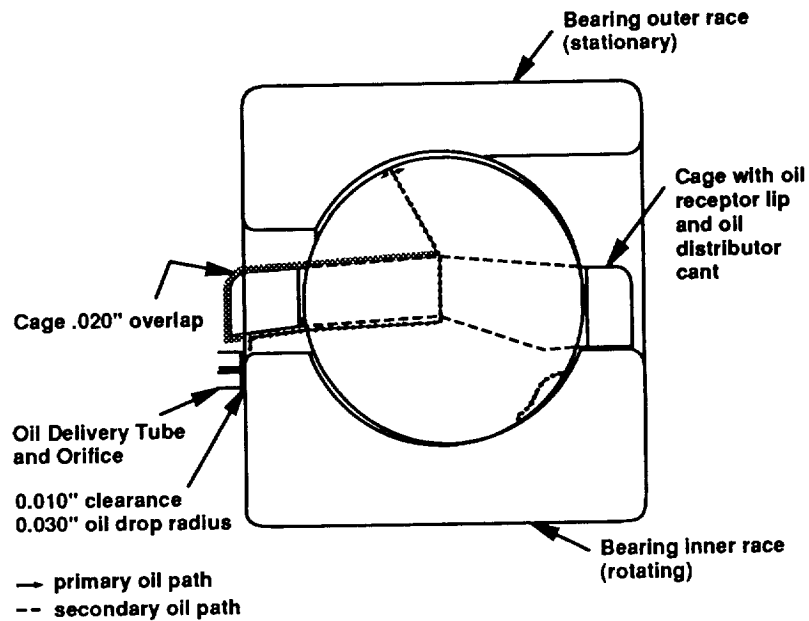


Figure 7. Previously-Developed Cage-Bearing Oil Distribution System (Option 1)

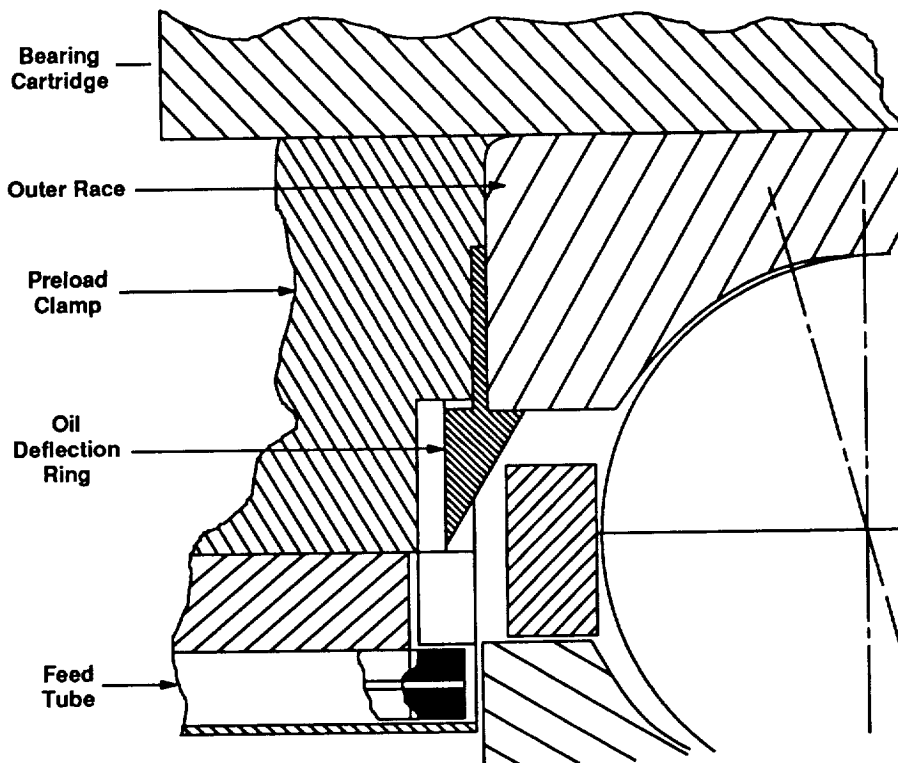


Figure 8. Oil Transfer (Option 2)

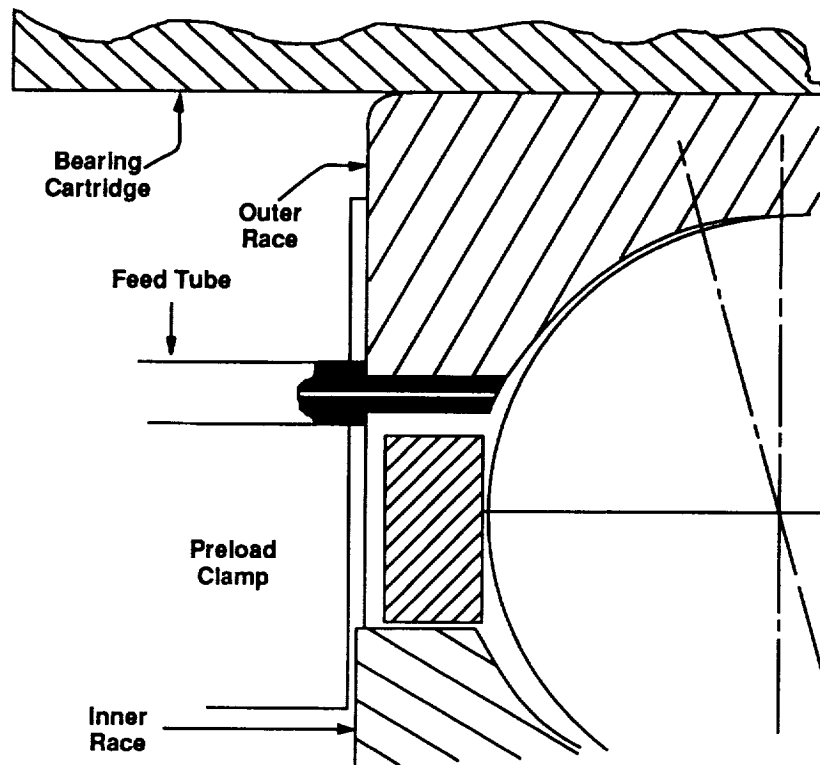


Figure 9. Oil Transfer (Option 3)

**DIRECT TO BALLS/HOLE IN OUTER RACE (OPTION 4)** – With Option 4 (Figure 10), oil flows through the feed tubes into the small annulus between the outer races of the bearings and the cartridge. The oil then flows through four 0.010-inch diameter holes in the outer race of each bearing from the annulus to the interior of the bearing at a point 0.002 to 0.004 inch from the ball.

This option would have two pressurized reservoirs for each bearing pair and provide redundancy for both the oil supply and the oil paths. Using this option, existing bearings and cartridges would have to be modified, but no new long-lead-time parts would have to be fabricated. The outer races of the bearings would have to have four diagonal holes added, as shown in Figure 10. The bearing cartridges would require the addition of an attachment hole to accept the feedtube from the pressurized reservoir and a small radial through hole to connect the attachment hole to the ID of the cartridge. This hole would coincide with the annulus at the interface between the bearings.

An analysis determined that the addition of these four holes would not compromise outer race load capacity. This was later confirmed by testing the modified bearings in an operational CMG.

**DUAL-PATH INSERTION TO RADIAL BEARING HOLES (OPTION 5)** – Option 5, shown in Figure 11, is identical to Option 4 except that the oil is injected through four radial (instead of diagonal) holes in the outer race of each bearing to the interior of the bearing.

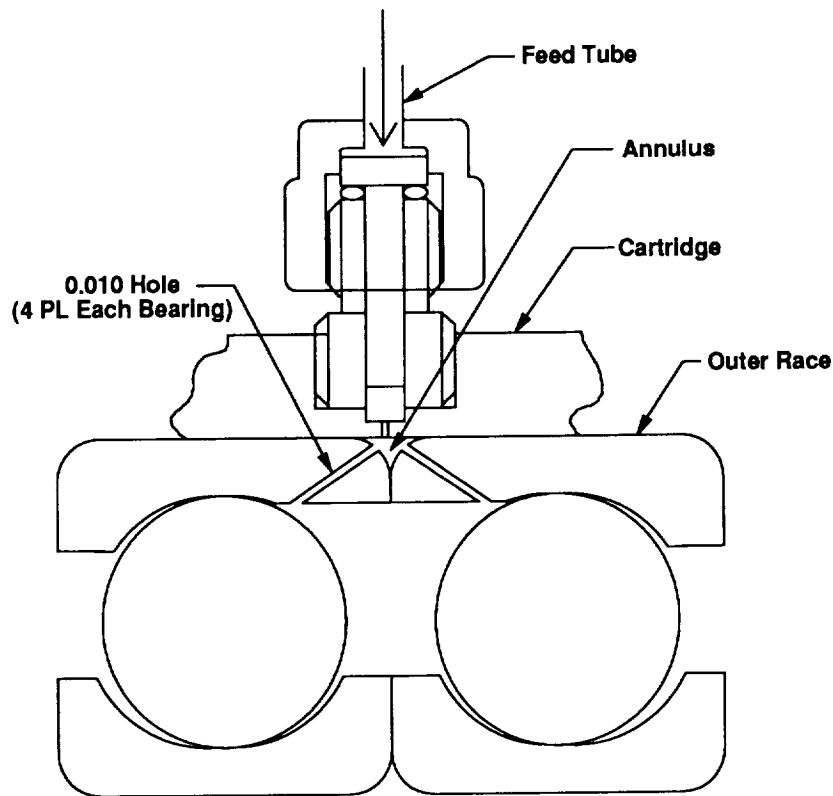


Figure 10. Oil Insertion (Option 4)

Using this design, one pressurized reservoir would supply oil to only one bearing. Option 5, like Option 4, would only require part modification instead of new long-lead-time parts. A machined groove was added to the ID of the bearing cartridge in two places. As with Option 4, an attachment hole was added to accept the feed tube from the pressurized reservoir, and a small radial through hole was electrodischarge-machined (EDMed) from the attachment hole to the ID of the cartridge. This hole intersected the annulus between the bearing OD and the machined groove on the ID of the bearing cartridge.

**OIL INSERTION/TRANSFER OPTION TESTING/SELECTION** – All tests were conducted using flight-designed 305 bearings and cages in the spindle test fixture (Figure 6). Oil was injected from the pressurized reservoir, using 125-second pulses at 5- to 10-second intervals. For all tests, the bearings and pressurized reservoirs were weighed before and after testing to determine the amount of oil transferred.

**OIL TRANSFER TESTING** – Options 2 and 3 (Figures 8 and 9) were the two methods initially considered for injecting oil into the bearings. These two methods were used for the transfer testing. Although a feed tube was used for the testing of Option 3, for a flight installation, a new preload bearing clamp would be designed to incorporate this oil injection feature.

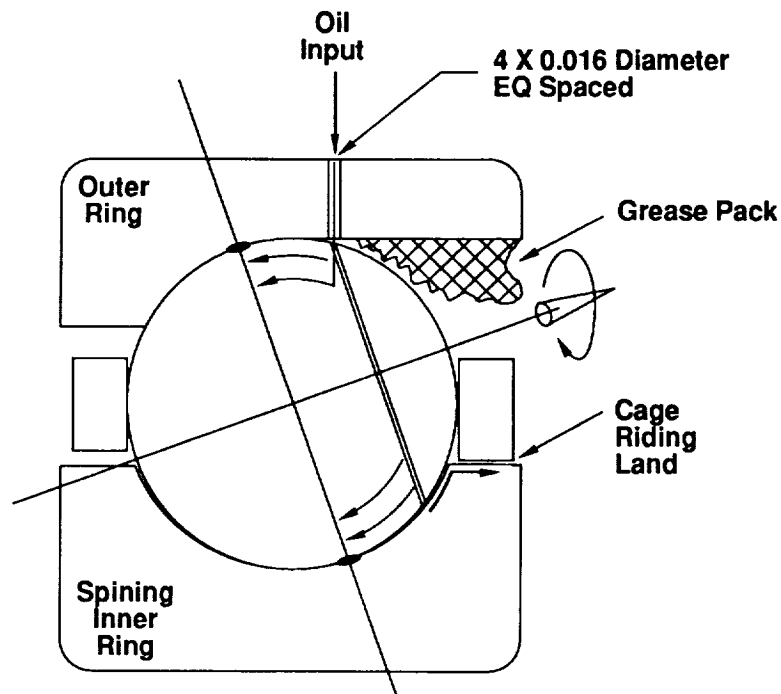


Figure 11. Nominal Bearing Geometry With Oil Holes (Option 5)

Numerous tests were run using both greased bearings and oil-only bearings. During the first five tests the bearings were injected with various quantities of oil while running in the test fixture. The bearings were weighed before and after testing to determine the amount of oil injected and the amount of transfer. Table 2 summarizes the results of the tests. The first four tests demonstrated consistent oil delivery to the outboard bearing but inconsistent transfer to the inboard bearing. Observations in earlier PLUS development tests indicated that oil would transfer from one bearing to the other; however, earlier tests were done using bearings with extended-lip cages (Figure 7). To determine if the type of cage was indeed a factor in oil transfer, two tests were run (No. 3 and No. 8) with an extended-lip cage. After the first five tests were performed there was concern that the weighing method, particularly with the greased bearings, was not an accurate enough determination of transfer because of the possibility of grease transfer between the bearings when they are separated. Since the testing needed to conclusively demonstrate transfer, the last three tests were done with greased bearing and the injected oil was impregnated with blue dye (Nitrofast Blue-2B®, manufactured by Sandoz Chemical). The blue dye demonstrated excellent delivery from the pressurized reservoir into the outboard bearing with all options, but practically no transfer from the outboard to inboard bearing. At this point the decision was made that each bearing would have to have its own direct oil injection to ensure proper bearing lubrication. During the testing of Option 2, the cage occasionally contacted the oil deflection ring. Based on this data and the unknown long-term effects that the ring could have on grease-cage coupling, this option was eliminated.

Table 2. PLUS Transfer Test Results

Test No.	Option No.	Oil/Grease	Oil Delivered (mg)	$\Delta$ O/B (mg)	$\Delta$ I/B (mg)	Transfer
1	3	Grease	88	Unknown	Unknown	Unlikely
2	3	Grease	200	+63	-104	Unlikely
3	2	Oil only	1115	+152	+39	Yes
4	3	Oil only	200	+46	+21	Yes
5	3	Oil only	127	+15.2	+22.2	Yes
6	3	Grease	188 (blue dye)	+95	-146	No
7	2	Grease	262 (blue dye)	-125	-204	No
8	3	Grease	500 (blue dye)	+153	-215	Some (visual indication)

**OIL INSERTION TESTING** – Oil insertion testing was performed using Options 3, 4, and 5.

The primary goal was to test the capability of the various options to transfer oil from the pressurized reservoir to the bearings and to select the best option for flight design.

Oil Insertion Tests (Option 3) – To simulate Option 3 oil insertion for testing, a feed tube machined to 0.025 inch thick was fabricated and installed between the cage and the bearing outer race to insert oil directly onto the balls. The feedtube was secured in the setup to hold it hard against the outer bearing race.

Four tests were run, three with greased bearings and one with oil-only bearings. During the first test the weight gain of the bearing was undetermined although there was visual evidence of oil insertion. During the second test 63 milligrams out of 200 were injected. The third test was conducted using an oil-only lubricated bearing. During the test, 46 out of 200 milligrams were injected onto the bearings. The final test was run with greased bearings, and the injected oil was impregnated with blue dye. The blue dye showed an excellent visual indication of delivery from the pressurized reservoir into the outboard bearing, with 95 out of 188 milligrams delivered to the bearing. As stated earlier, bearing weights were not accurate due to grease transfer between bearings.

All four tests demonstrated consistent oil delivery into the bearings. Because of the results of the other testing and analysis as well as manufacturing considerations, the decision was made to eliminate this option.

Oil Insertion Tests (Option 4) – The test-cartridge was modified to accept a feed tube similar to the flight concept, and a set of test bearings was modified as shown in Figure 10. For these tests, one pressurized reservoir was used to supply both bearings.

Six tests were conducted using this option with oil-only bearings. For the tests, the bearings were run in a vacuum test fixture until the drag torque increased. At that time oil was injected into the bearing. In all of the tests, the drag torque immediately decreased.

Four tests were run using grease-lubricated bearings. The tests were run for four to six hours and then injected with 50 to 100 milligrams of oil (with blue dye). After the first test, the visual inspection indicated that oil flowed through only one of the eight holes (four holes in each bearing) in the two bearings. During the other three tests, various combinations of holes were manually plugged with grease, and oil quantities from 56 to 106 milligrams were injected. The results of the post-test visual inspections always showed that the

oil followed the lowest resistance path, which resulted in only one of the bearings receiving the oil.

The blue dye proved to be an excellent visual indication of oil insertion. It was observed that grease covered the oil outlet holes during every test, though the oil could readily unplug a grease-filled hole. Once one hole was unplugged, the others would not unplug, and oil would go into only one bearing.

The conclusion of this testing was that each pressurized reservoir could inject oil into only one of the two bearings. Because of this result and the manufacturing difficulties associated with the diagonal holes, Option 4 was abandoned.

**Oil Insertion Test (Option 5)** – The test bearings and cartridge were modified as described in the Dual-Path Insertion to Radial Bearing Holes (Option 5) paragraph. Three tests were run using greased bearings. For all of the Option 5 tests, the injected oil was impregnated with blue dye.

During the first test, 129 milligrams of oil were injected into the outboard bearings and 93 milligrams into the inboard bearings. After the test, the visual inspection indicated excellent oil transfer into both bearings and excellent distribution of the injected oil. During the second test, 74 milligrams of oil were injected into the outboard bearing and 39 milligrams into the inboard bearing. After the test, the visual inspection indicated excellent oil transfer and distribution in the outboard bearing but very little in the inboard bearing; this was due to the small amount injected. It takes 25 to 30 milligrams of oil to fill the annulus in the bearing cartridge before any oil can flow into the holes in the bearings.

The third test was designated the long-term test. For this test, the bearings were run for 14 days prior to oil injection. The purpose of this test was to determine if dry grease would effect the oil injection. The test cartridge used had no internal oil reservoirs and an open-labyrinth seal to allow the grease to dry out quickly. After the long run time, 108 milligrams of oil were injected into the outboard bearing and 84 milligrams into the inboard bearing. After the test, the visual inspection indicated excellent oil transfer and distribution in both bearings.

Several oil insertion tests had been performed on oil-only lubricated bearings to determine the system's ability to reduce drag torque. During these tests, the bearings were run until the drag torque and temperature increased and then various quantities of oil were injected. In all the tests, the drag torque, temperature, and observed cage instability decreased immediately after oil injection. These tests determined that 20 to 40 milligrams of oil injected per bearing were usually all that was required to restore drag torques to normal.

All the tests using this option demonstrated very consistent oil transfer into the bearings. Because of the results of these tests, the earlier testing, analysis, and manufacturing considerations, the decision was made to select Option 5 for the PLUS oil insertion technique.

**OIL DISTRIBUTION – DYED OIL INJECTION** – Other tests were conducted using the same test setup to determine the effects of injecting oil into a bearing with oil-depleted grease. Another objective was to determine the amount of oil that can be added to a set of bearings without causing an increase in the drag torque.

A set of greased bearings was starved by operating them at elevated temperatures in an open cartridge to evaporate 40 to 60 percent of the original oil. They were then installed

in the test setup and various quantities of oil were injected into the bearings. No significant changes in drag torque were noted following oil injection.

After the tests, a visual examination showed that the grease was wetted throughout, all surfaces showed good oil distribution, cages went from dry (before injection) to well oiled (after injection), and up to 40 milligrams could be added to a bearing without measurable change in drag torque.

**FLOODING** – Another test was conducted using the same test setup to determine the effects of injecting an entire reservoir full of oil (~2 grams) into a set of fresh bearings. This will simulate a worse-case failure of the PLUS (valve-open failure) early in the bearing life.

After running the bearings for about 40 hours the drag torque had stabilized at 2.30 to 2.50 oz-in. The reservoir valve was opened and remained open until all of the oil was discharged. The torque increased immediately but then slowly decreased until it returned to normal levels. This test showed that a valve-open failure will not cause long-term, high drag torques or other observable problems.

**FIXTURE TEST CONCLUSION** – The conclusions of these tests are: oil added to greased bearing will successfully migrate to the bearing critical areas; oil can be added to the bearings in small quantities without increasing drag torque; and oil added in large quantities does not cause long-lasting effects on bearing performance. Option 5 is the best method for injecting oil into bearings.

#### **LIFE TESTS**

Oil added to distressed bearings results in recovery of performance. This was demonstrated many times with the bearing spindle test fixture and has also been demonstrated in life tests. An RWA life test stopped after one year due to high bearing drag torques. After an addition of 40 to 50 mg of oil, the test was restarted and ran for three years. The unit was still operating normally when the test was terminated due to program consideration.

An accelerated life test on a momentum device was stopped at 67 percent of expected life because of abnormal bearing behavior (high drag torques and temperatures). The bearing had 50 mg of oil added to each bearing. The test was restarted and has run normally for an additional 17 percent of life. The current plan is to continue to add oil at fixed intervals to extend the life of the test well beyond its original planned duration.

#### **PRESENT/FUTURE DEVELOPMENT**

The current PLUS has been designed, built, and flight qualified.

The design of the next generation PLUS is in progress (Figure 12). Some of the features of the new design is a size-optimized design with full mechanical and electrical redundancy and constant output (not dependent on pressure, temperature, etc). The new design will have timed injection combined with manual override capability to provide a degree of autonomy.

Currently, there are outstanding proposals to install the PLUS on a small CMG and an RWA flight program.

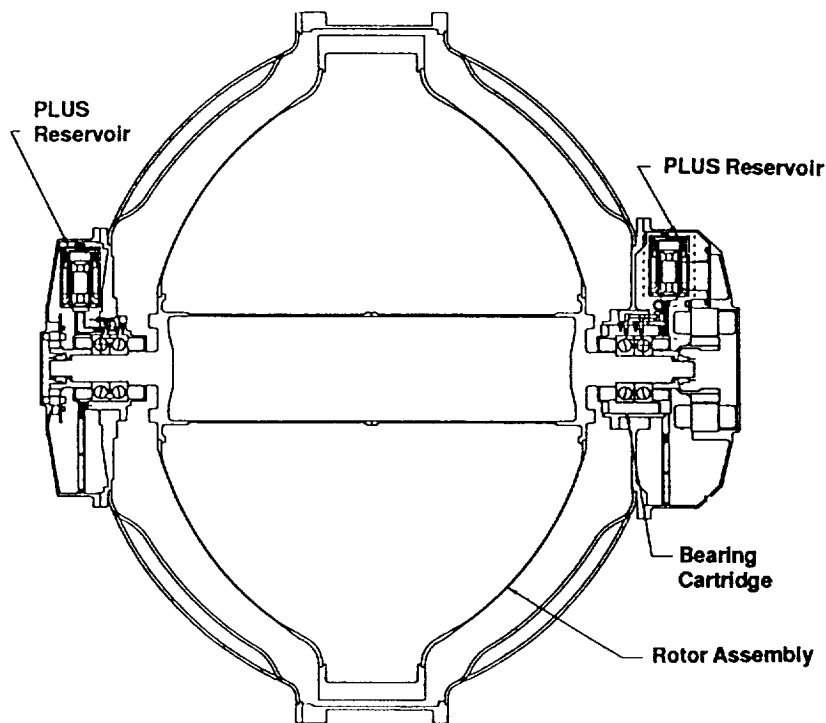


Figure 12. Typical Installation.

Also under development with IR&D funding is a lubricant monitor that senses the lubricant requirement and a control system that will command oil injection at the proper time, based on sensor output. This monitor and control system would be used with the present PLUS for an autonomous active lubrication system. (The present autonomous system will inject a fixed amount of oil at predetermined, but variable, intervals.)

#### **SUMMARY AND CONCLUSIONS**

The most significant observation in most bearing failures is lubricant degradation. Maintenance of adequate lubrication is the key to extended life. A well-lubricated bearing will run an extremely long time (fatigue life limited only).

The PLUS has shown to be an effective method of extending the life of grease-lubricated bearings well beyond passive system capabilities by adding oil. Tests have shown that the PLUS delivers oil to the critical bearing areas on demand. The PLUS is mission flexible with its full autonomy and has the capability to alter the lubrication cycle in response to unexpected demands. The PLUS can prevent anomalies or intervene if necessary to mitigate unexpected problems. It also has end-to-end ground test capability to verify operation before flight use. The PLUS provides lubrication life that is basically limited only by reservoir size.